FAUST

Design of a Portable Pneumatic Riveter

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DESIGN OF A PORTABLE PNEUMATIC RIVETER

BY

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THESIS

FOR THE

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IN

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Per Alexander Faust

ENTITLED Design of a Portable Pneumatic Riveter

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science

in Mechanical Engineering.

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Instructor in Charge

APPROVED: C. R. Richards

HEAD OF DEPARTMENT OF Mechanical Engineering.

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CHARTS AND PLATES.

DESIGN

OF A

PORTABLE PNEUMATIC RIVETER.

INTRODUCTION.

The object of this thesis is to design a portable pneumatic riveter fulfilling the specifications given below.

Portable pneumatic riveters are used in structural steel and boiler construction on account of the ease with which power can be supplied to them and on account of the great amount of work of good quality that can be turned out by them.

These machines are of the C frame type and differ only in the toggle, or lever motion. They are practically all hand operated, and are so designed that the maximum pressure is obtained at the end of the stroke.

SPECIFICATIONS.

1. Capacity - The machine shall have a capacity of 3/4 inch rivets on boiler work.

2. Depth of Throat - The throat shall have a depth so that a 3/4 inch rivet can be set 24 inches from the edge of the plate.

3. Stroke of Plunger - The plunger shall have a stroke of 2 1/2 inches.

4. Distance between Jaws - The distance between the jaws shall be 12 inches.

5. Style of Toggle - The machine shall be equipped with the "Hanna toggle".

6. Ratio of Piston travel to Plunger travel - The ratio of travel between the piston and plunger shall be 4 to 1.

7. Speed of Piston - The piston shall have a speed of 20 feet per minute.

8. Speed of Plunger - The plunger shall have a speed of 5 feet per minute.

9. Material - The frame shall be made of steel casting.

10. Allowable Stress - The stresses in the frame shall not exceed 16000 pounds per square inch.

11. Method of Control - The machine shall be controlled by a hand lever connected to the valve.

12. Air Pressure - The machine shall be designed to operate on an air pressure of 80 pounds per square inch.

13. Distance of Die above floor - The die table shall be 30 inches above the floor.

CHAPTER I.

SELECTION OF STYLE OF TOGGLE.

The specifications call for a riveter to set 3/4 inch rivets on boiler work, hence, the first step taken was to find the pressure required to do this work.

This pressure cannot be determined theoretically. However, it has been determined experimentally by manufacturers of riveters so that the data can be obtained from trade catalogues and standard handbooks. The information in this particular instance was obtained from Kent's Mechanical Engineer's Hand Book (8th edition) and catalogue Number 10 of the Chester B. Albree Iron Works. The latter gives the following rule

$$P = 150000\,A \quad . . . . (1)$$

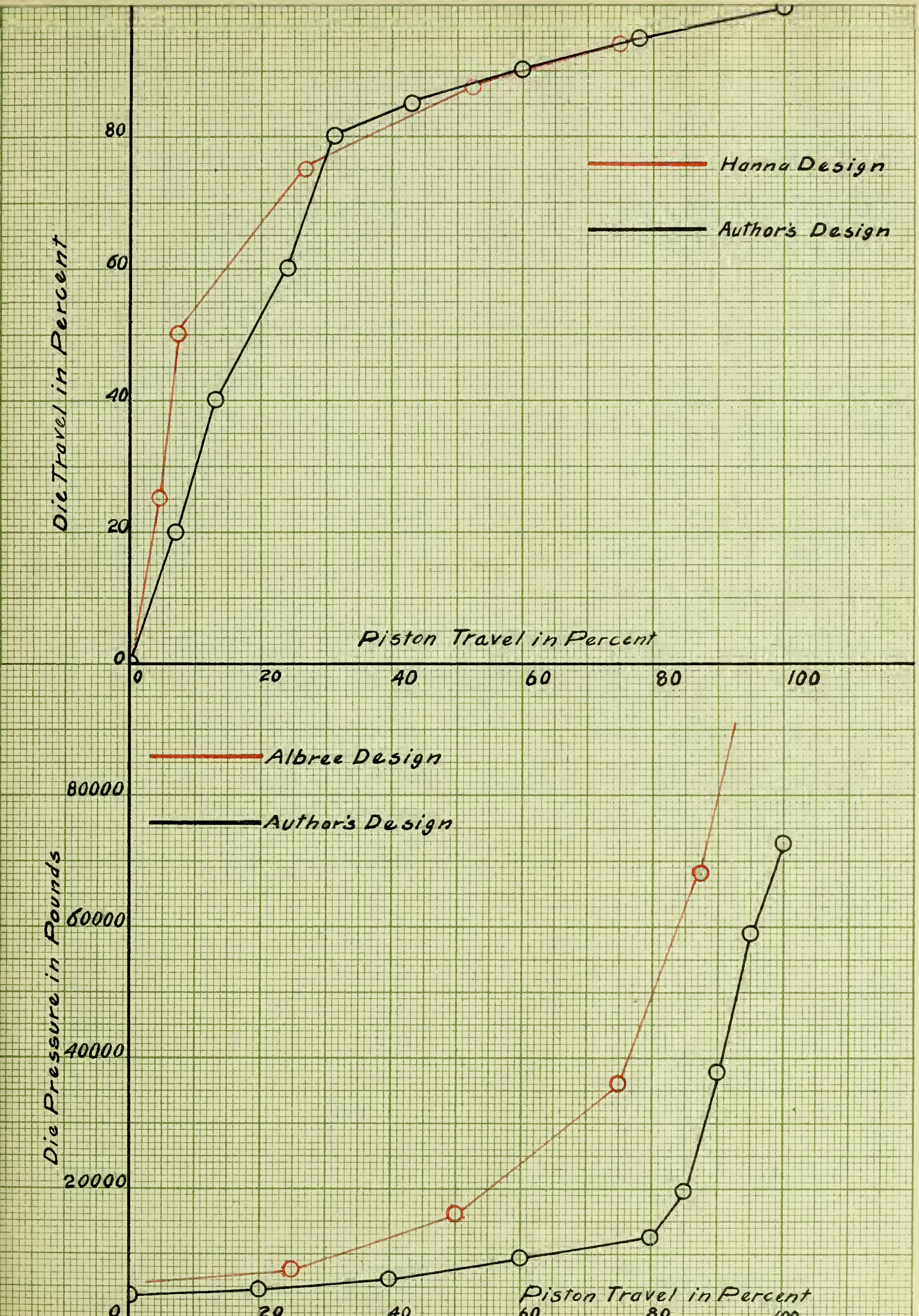
in which P = pressure required

and A = area of rivets in square inches.

Hence $P = 150000 \frac{(0.75)^2}{4} = 66270$ pounds.

This value agrees very closely with that given by Mr. Kent who gives 66000 pounds. This value will be used in subsequent calculations.

Obviously this pressure is not required thruout the whole


Hanna Design
Author's Design
Die Travel in Percent
80
60
40
20
0
Piston Travel in Percent
0
20
40
60
80
100
Albree Design
Author's Design
Die Pressure in Pounds
80000
60000
40000
20000
0
Piston Travel in Percent
20
40
60
80
100

stroke, but merely at the end of the stroke so as to give the final "squeeze" to set the rivet. But since no means were at hand for determining the variation of the required pressure at every point of the stroke, a table, given in the above mentioned catalogue was made use of, The pressures given are plotted against the stroke on page 4.

For additional information and in order to determine how closely rival manufacturers agree on this point the die travel was plotted against the piston travel of a Hanna riveter. This curve (shown on page 4) gave what the machine was capable of doing at any point of the stroke. Therefore, a machine designed that was capable of exerting a pressure that varied according to the Hanna or Albree machine would do the work satisfactorily. This reduces the problem to a selection between the simple lever motion used by the Albree Iron Works, John F. Allen, and several other makers and the more complicated Hanna motion. The Hanna toggle motion was selected more for the study of kinematics that it would afford than for any especial advantage that it possesses over any of the other types.

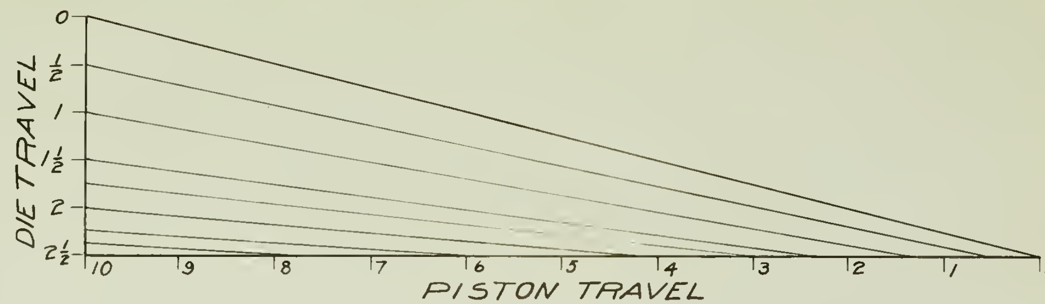


Fig. 3

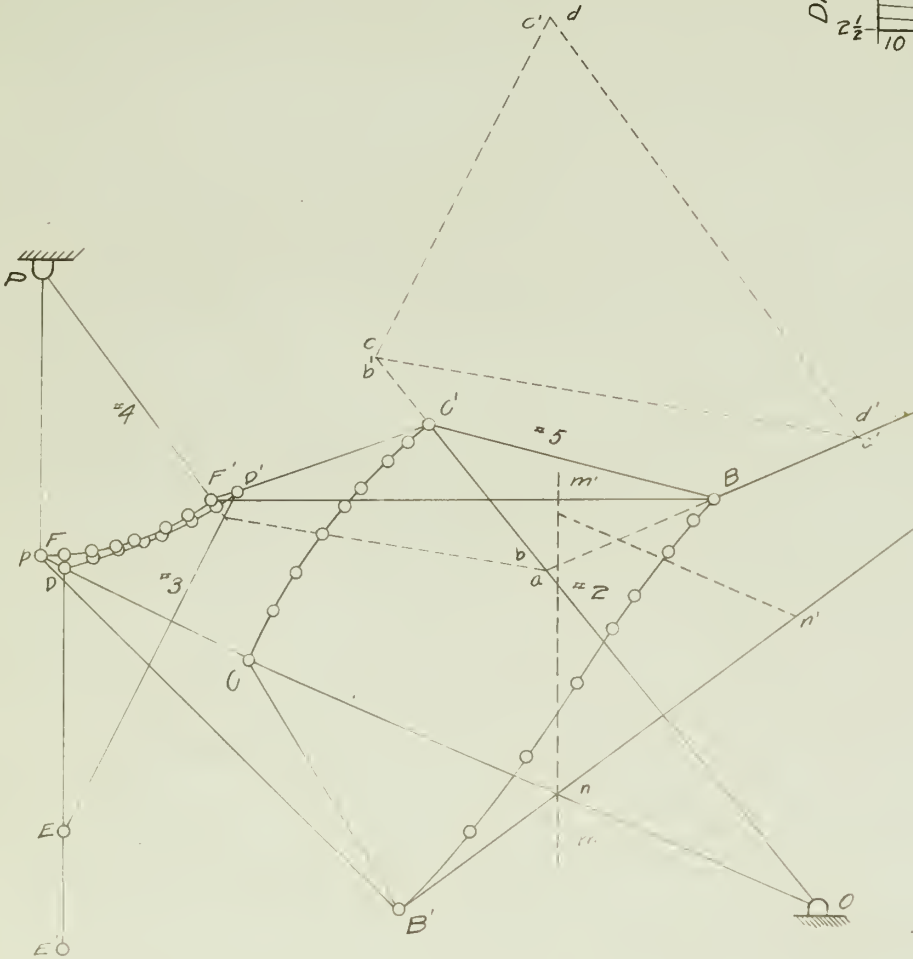


Fig. 1

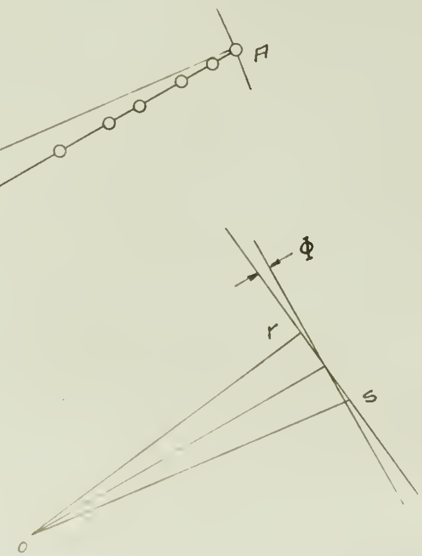


Fig. 2

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M.E. 33
PORTABLE RIVETER
KINEMATIC & STRESS DIAGRAMS
SCALE ½"=1", 1"=1", 1"=1000 lb.
MAY 3, 1912
P.U. Faust

CHAPTER II.

Kinematics of the Toggle Motion.

The toggle motion consists of the following five links as shown in Fig. 1, page 6 . The connecting rod — link 1; the toggle guide —link 2; the lower link - link 3; the toggle link - link 4; and the toggle lever - link 5.

The connecting rod connects the toggle lever to the piston. The toggle guides connect the toggle lever to the frame and determine one direction of motion of the lever. The lower link is the pendulum that transmits the motion from the toggle lever to the plunger. The toggle link is the upper link and forms a connection between the frame and the toggle lever. It determines the second direction of the lever motion. The toggle lever is a quaternary link and is the link thru which the pressure is exerted.

As the piston moves from A to A', Fig. 1, page 6 one end of the link 1 moves along the path AA'. Now, since the links 1, 2, and 3 are all connected to link 5, and since 2 must move about the point O and 4 about the point P, links 2 and 4 describe the arcs CC' and FF' respectively. This constrains the motion of link 5 so that the joint (1-5) moves along the path BB' and the joint (3-5) along the path DD' thus forcing the plunger down along the path EE'.

CHAPTER III.

Design of Frame and Details.

In designing any part of a machine it is necessary to analyze all the forces acting upon it and to determine the maximum stresses caused by these forces. However, it is not always practical to design a machine so that all parts of the machine have equal stresses because of difficulties of construction. It is better to design so that all parts are symmetrical and so that the parts can be easily constructed. It is necessary, however, to investigate the stresses in all parts so that none of them exceed the allowable safe value.

In a C frame it is customary to take four sections as follows

(1) One along the horizontal.

(2) One at 45 degrees.

(3) One over a vertical line near the back of the throat.

(4) One on a vertical line near the head of the machine.

FORCE ANALYSIS.

The forces acting on the frame are as follows:-

(1) The force P acting along the center line of the frame.

(2) The bending action of P which acts with a lever arm e (measured from the center line to the gravity axis of the section). This causes a tension at T and a compression at C.

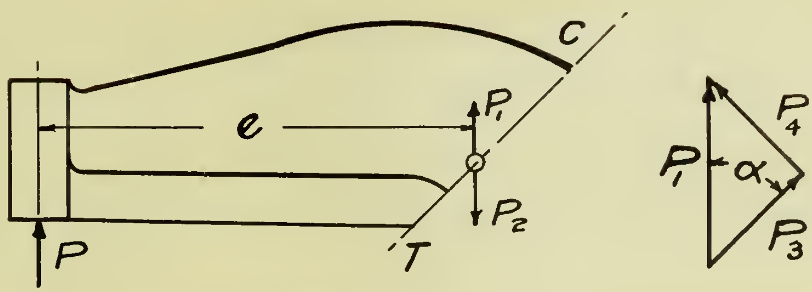


Fig. 1.

These forces are as follows

$$S_t = \frac{Pec_t}{I} \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

$$S_c = \frac{Pec_c}{I} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

where S_t = tensile stress.

S_c = compressive stress.

c_t = distance of the outermost fiber in tension from the gravity axis.

c_c = distance of the outermost fiber in compression from the gravity axis.

I = Moment of Inertia of the section.

P = the pressure.

(3) Equal and opposite forces P_1 and P_2 may be introduced at the center of gravity O without disturbing the equilibrium. The forces now acting upon the frame are the external

couple Pe and the external force P_1. The force P_1 may be resolved into two components, P_3 and P_4, one along the line TC and the other at right angles thereto. The component along TC causes a pure shear and is given by the equation

$$S_s = \frac{P \cos \alpha}{A} \quad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

where S_s = the shearing stress.

P = pressure exerted on the frame.

α = the angle that the component along TC makes with the force P_1.

A = the area of the section.

The component at right angles to TC causes a tension and is given by the equation

$$S_t' = \frac{P \sin \alpha}{A} \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

where S_t' = tensile stress and the other symbols are as before.

To determine the resultant maximum stresses in the section use the formulas given in Merriman's Mechanics of Materials, page 225

$$\text{Max. } S_t = \frac{S_t + S_t'}{2} + \sqrt{S_s^2 + \left[\frac{S_t + S_t'}{2}\right]^2} \quad \ldots\ldots \quad (6)$$

$$\text{Max. } S_c = \frac{S_c - S_t'}{2} + \sqrt{S_s^2 + \left[\frac{S_c - S_t'}{2}\right]^2} \quad \ldots\ldots \quad (7)$$

$$\text{Max. } S_s = \sqrt{S_s^2 + \left[\frac{S_t + S_t'}{2}\right]^2} \text{ or}$$

$$\sqrt{S_s^2 + \left[\frac{S_c - S_t'}{2}\right]^2} \quad \ldots\ldots\ldots \quad (8)$$

where S_t = maximum tensile stress.

S_c = maximum compressive stress.

S_s = maximum shearing stress.

HORIZONTAL SECTION.

Assuming an I section having the general proportions indicated in Fig. 2 and calculating the properties of the section the values tabulated in Table I were obtained.

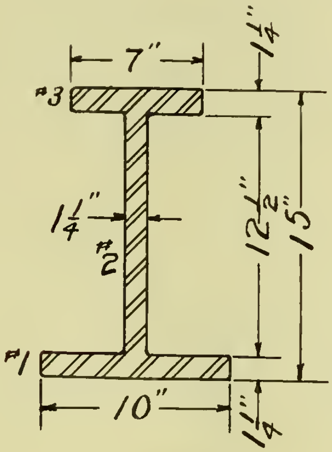


Fig. 2.

TABLE I.

No	Area sq. in.	Mom. Arm. in.	Mom.	C_t	C_c	Icg.	h	Ah^2	I	$\frac{I}{c}$ T_c	$\frac{I}{c}$ C_c
1	12.5	0.625	7.8	6 .	8 .	1.625	6.175	475	475.63	1	1
2	15.6	7.500	117.0	8	2	204	0.7	7.65	211.65	7	4
3	8.75	14.375	125.8	0	0	1.14	7.575	501	502.14	5	5
	36.85		250.6						1190.42		

Substituting the values of Table I in formulas (2) to (8) inclusive the following values were obtained

$$P = 66000 \text{ lb.} \qquad e = 31.3 \text{ in.}$$

$$S_t = \frac{66000 \times 31.3}{175} = 11800 \text{ lb./ sq. in.}$$

$$S_c = \frac{66000 \times 31.3}{145} = 14250 \text{ lb./ sq. in.}$$

$$S_t' = \frac{66000}{36.85} = 1790 \text{ lb./ sq. in.}$$

$$\text{Max. } S_t = 11800 + 1790 = 13590 \text{ lb./ sq. in.}$$

Max. S_c = 14250 + 1790 = 16040 lb./ sq. in.

These maximum values are within the limits given in the specifications and the section therefore can be used.

45 DEGREE SECTION.

The dimensions shown in Fig. 3 were assumed and the properties of the section were calculated and the results were tabulated in Table II.

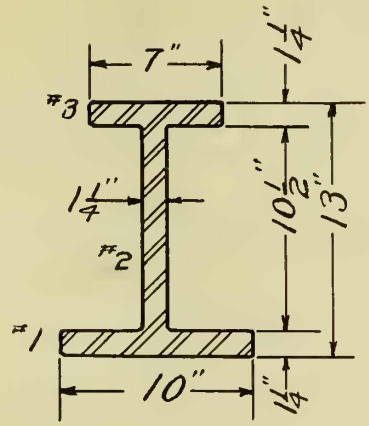


Fig. 3.

TABLE II.

<table>
<tr><th rowspan="2">No</th><th rowspan="2">Area sq. in.</th><th rowspan="2">Mom. Arm. in.</th><th rowspan="2">Mom.</th><th rowspan="2">C_t</th><th rowspan="2">C_c</th><th rowspan="2">Icg.</th><th rowspan="2">h</th><th rowspan="2">Ah^2</th><th rowspan="2">I</th><th colspan="2">$\frac{I}{c}$</th></tr>
<tr><th>T_c</th><th>C_c</th></tr>
<tr><td>1</td><td>12.5</td><td>0.625</td><td>7.8</td><td rowspan="4">5.85</td><td rowspan="4">7.15</td><td>1.625</td><td>5.225</td><td>326</td><td>327.625</td><td rowspan="4">141.1</td><td rowspan="4">115.4</td></tr>
<tr><td>2</td><td>13.1</td><td>6.5</td><td>85.2</td><td>120.5</td><td>0.65</td><td>5.5</td><td>126.</td></tr>
<tr><td>3</td><td>8.75</td><td>12.375</td><td>108.2</td><td>1.14</td><td>6.525</td><td>372</td><td>373.14</td></tr>
<tr><td></td><td>34.35</td><td></td><td>201.2</td><td></td><td></td><td></td><td>826.765</td></tr>
</table>

The values tabulated in Table II were substituted in the formulas (2) to (8) inclusive, and the following stresses were obtained.

P = 66000 lb. e = 27.16 in.

$$S_t = \frac{66000 \times 27.16}{1411} = 12700 \text{ lb./ sq. in.}$$

$$S_c = \frac{66000 \times 27.16}{115.4} = 15500 \text{ lb./ sq. in.}$$

$$S_t' = \frac{66000 \times 0.707}{34.35} = 1360 \text{ lb./ sq. in.}$$

$$\text{Max. } S_t = \frac{12700+1360}{2} + \sqrt{(1360)^2 + \frac{(12700+1360)^2}{4}}$$

$$= 14180 \text{ lb./ sq. in.}$$

$$\text{Max. } S_c = \frac{15500+1360}{2} + \sqrt{(1360)^2 + \frac{(12700+1360)^2}{4}}$$

$$= 14270 \text{ lb./ sq. in.}$$

$$\text{Max. } S_s = \sqrt{(1360)^2 + \frac{(12700 + 1360)^2}{4}}$$

$$= 7150 \text{ lb./ sq. in.}$$

$$\text{Max. } S_s = \sqrt{(1360)^2 + \frac{(15500 - 1360)^2}{4}}$$

$$= 7200 \text{ lb./ sq. in.}$$

These stresses are well within the safe limits and hence the section assumed has the proper dimensions.

VERTICAL SECTION.

The dimensions shown in Fig. 4 were assumed and as in the case of the previous sections the properties were calculated. The results of these calculations are shown in Table III.

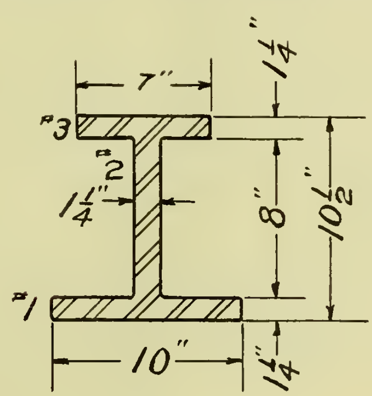


Fig. 4.

TABLE III.

No	Area sq. in.	Mom. Arm. in.	Mom.	C_t	C_c	Icg.	h	Ah^2	I	$\frac{I}{c}$ T_c	$\frac{I}{c}$ C_c
1	12.5	0.625	7.8	4 .	5 .	1.625	4.055	206	207.625	1	9
2	10	5.25	52.5	6	8	53.3	0.57	3.24	56.54	1	1
3	8.75	9.875	86.5	8	2	1.14	5.195	266	267.14	3 .	3
	31.25		146.8						531.305	5	

Substituting these values in formulas (2) to (8) inclusive, the following values were obtained. These are within the safe limits and, hence, the section selected is of the proper size.

$$P = 66000 \text{ lb.} \qquad e = 19.5 \text{ in.}$$

$$S_t = \frac{66000 \times 19.5}{113.5} = 11340 \text{ lb./ sq. in.}$$

$$S_c = \frac{66000 \times 19.5}{91.3} = 14100 \text{ lb./ sq. in.}$$

$$S_s = \frac{66000}{31.25} = 2100 \text{ lb./ sq. in.}$$

$$\text{Max. } S_t = \frac{11340}{2} + \sqrt{(2100)^2 + \frac{(11340)^2}{4}}$$

$$= 5670 + 6040 = 11710 \text{ lb./ sq. in.}$$

$$\text{Max. } S_c = \frac{14100}{2} + \sqrt{(2100)^2 + \frac{(14100)^2}{4}}$$

$$= 14430 \text{ lb./ sq. in.}$$

$$\text{Max. } S_s = \sqrt{(2100)^2 + \frac{(14100)^2}{4}} = 7350 \text{ lb./ sq. in.}$$

CYLINDER.

In order to calculate the diameter of the cylinder it is necessary to work from the maximum required pressure i.e. from the position where the rivet is set.

In this position the links 3 and 4 are parallel. The stress

and direction of link 3 is known and the direction of the links 1, 2, and 4 is known. Hence, combining 1 and 2 and taking moments about the point P the magnitude of the resultant of 1 and 2 is determined. From this resultant the stress in the connecting rod is obtained. This is shown graphically by the dotted lines in Fig. 1 page 6.

In this diagram mm' is the stress in the resultant of 1 and 2, and nn' is the stress in the connecting rod, using a coefficient of friction $\mu = 0.1$. The pressure required on the piston is therefore

$$P = 1000 \times 2.93 = 2930 \text{ pounds.}$$

Assuming an air pressure of 80 pounds per square inch

$$P = \frac{\pi d^2 p}{4}$$

where P = required pressure in pounds.

p = available pressure in pounds per square inch.

d = diameter of the piston.

from which,

$$d = \sqrt{\frac{4P}{\pi p}} \quad \ldots\ldots\ldots\ldots \quad (9)$$

Substituting the values of P and p in (9) we get

$$d = 6.83 \text{ inches.}$$

In order to allow for a drop of pressure and leakage, the cylinder will be made 7 1/2 inches in diameter.

Allowing 0.25 inches for reboring and assuming an outside diameter of 9 1/2 inches, then

$$K = \frac{9.5}{7.75} = 1.225$$

where K is the ratio of the outside to the inside diameter.

Then, from Cook's formula (London Engineering, December 15, 1911. page 786)

$$P' = f\left[\frac{K^2 - 1}{K^2 + 1}\right] \quad \ldots\ldots\ldots\ldots\ldots \quad (10)$$

where P' = the rupture pressure

and f = the ultimate strength.

Substituting the known values of K and f we get

$$P' = 18500\left[\frac{(1.225)^2 - 1}{(1.225)^2 + 1}\right] = 3700 \text{ pounds per square inch.}$$

which is a safe value to use.

PISTON.

In the design of the piston, formulas are of very little use, and the good judgement of the designer plays a very important part. The piston must not be too short or it might cause undue wear on the cylinder or the piston itself. The back wall of the piston can be designed for strength and for that purpose the following formula taken from Merriman's Mechanics of Materials page 410 will be used.

$$T = \frac{3}{4}\left[\frac{r}{d}\right]^2 R \quad \ldots\ldots\ldots\ldots\ldots \quad (11)$$

where T = tensile stress

r = radius of the piston wall

R = the pressure in pounds per square inch.

d = thickness of the wall.

Assuming a thickness of 3/4 inch

$$T = \frac{3}{4}\left[\frac{2.5}{0.75}\right]^2 80 = 665 \text{ pounds per square inch.}$$

This is a safe value for the material used in making the piston.

CYLINDER BACK HEAD.

Using the same formulas as above, and assuming a thickness of 3/4 inches, we get

$$T = \frac{3}{4}\left[\frac{3.875}{0.75}\right]^2 80 = 1600 \text{ lb./sq. in.}$$

This stress is somewhat higher than that found for the piston but is still on the side of safety, hence the thickness 3/4 inch will be used.

DETERMINATION OF THE STRESSES IN THE LINKS.

The stresses in the links were determined graphically because it was evident that they would vary for different positions of the piston.

Since the pressure on the piston and the coefficient of friction were known the force polygon ors shown in Fig. 2, page 6 , was constructed. This gave the stress in link 1 for that position. Hence the stress in one link and the direction of all the links were known. Therefore, applying the theorem on the resolution of Non-Concurrent forces, the stresses in the links 2, 3, and 4 were determined. These stresses are represented in Fig.1, page 6.

aa' = stress in link 1

bb' = stress in link 2

cc' = stress in link 3

dd' = stress in link 4

Repeating these constructions for various positions of the piston the following stresses shown in Table IV were obtained. To show more clearly the way these stresses vary Figs. 4, 5, 6, and 7, pages 34 and 35 were plotted.

TABLE IV.

Position of Die.	Link 1	Link 2	Link 3	Link 4	Die Pressure.
0	3550	2850	4000	5480	3600
1/2"	3550	2920	4950	6220	4550
1"	3550	3300	6450	7700	6050
1 1/2"	3530	3800	9560	8830	9200
2"	3530	4050	12500	13750	12250
2 1/8"	3530	4530	20000	20400	19800
2 1/4"	3510	3650	36100	38100	38000
2 3/8"	3510	380	59200	57120	59000
2 1/2"	3510	3100	72500	69400	72500

DESIGN OF THE CONNECTING ROD AND TOGGLE GUIDES.

The connecting rod and the toggle guides will be treated as columns and for the design of these parts the straight line formula for columns given by the following equation will be used.

$$S = 16000 - 70\frac{l}{r} \quad . . . . (12)$$

where S = the stress in the member

l = length of the rod

r = least radius of gyration.

$$S = \frac{P}{a} \quad . . . . (13)$$

where P = pressure = 3550 pounds

a = required area at root of thread.

Assume a rod 1 inch in diameter hence, the diameter at the root of the thread, assuming 8 threads per inch, is 0.81 inches. The least radius of gyration for the circular cross section is

0.2025 inches.

Substituting the above values in (12) we have

$$\frac{P}{a} = 9775 \text{ pounds.}$$

from which the required area

$$a = 0.363 \text{ sq. in.}$$

Comparing this calculated area with the area 0.52 square inches of the assumed rod it is seen that there is a margin of safety.

CONNECTING ROD PIN.

Assume a connecting rod bearing 2 1/2 inches wide as shown in Fig. 5. Then a bearing 1 1/4 inches wide is required on each side of the piston.

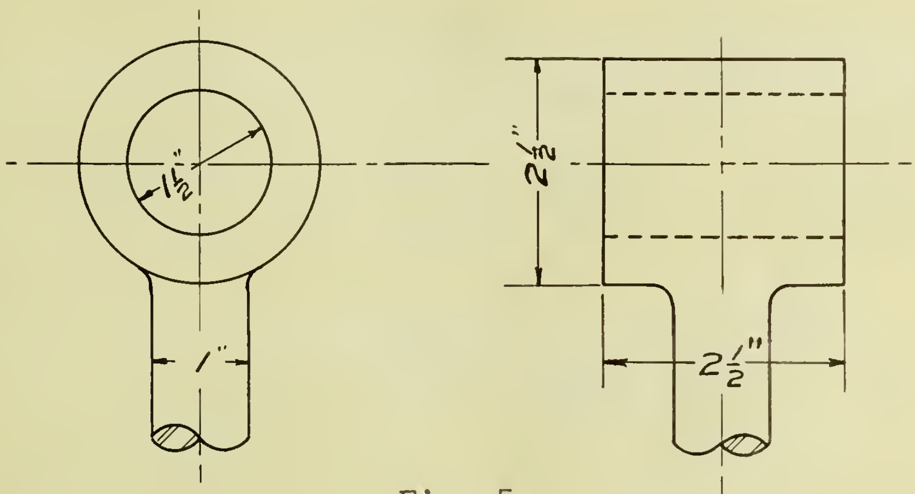


Fig. 5.

Taking moments about the center of the piston bearing

$$M = \frac{3550}{2}\left[0.625 + 1.25\right] = 3350 \text{ lb. in.}$$

Assuming a tensile strength of 12000 pounds per square inch then from the formula

$$\frac{M}{S} = \frac{I}{c} = \frac{\pi d^3}{32} \quad \ldots\ldots\ldots\ldots\ldots \quad (14)$$

we get by substituting the known values of M and S in (14)

$$d = 1.419 \text{ inches.}$$

Selecting a standard size we will increase this to 1 1/2 inches.

BEARING.

The bearing proper must be investigated for both crushing and shearing. The crushing stress is obtained by dividing the force P by the projected area, hence we have

$$S_c = \frac{P}{a} = \frac{3550}{2 \times 1.25 \times 1.5} = 950 \text{ lb./sq. in.}$$

This is a safe value for this class of bearing. To obtain the shearing stress divide the force P by the total area in shear which gives

$$S_s = \frac{2 \times 3550}{(1.5)^2} = 1000 \text{ lb./sq. in.}$$

This is rather low for the material used but it would not be advisable to decrease the dimensions of the pin, as the crushing stress would run up too high, hence the dimensions as chosen above are satisfactory.

TOGGLE LEVER GUIDES.

Assume a cross section for these guides as shown in Fig. 6.

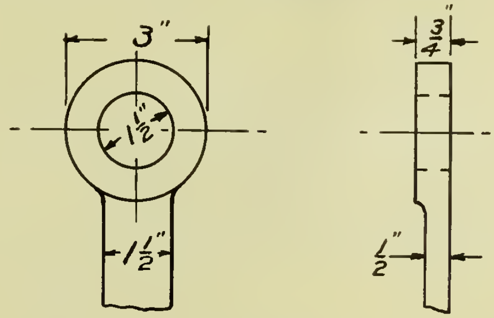


Fig. 6.

For the rectangular cross section shown, the least radius of gyration is 0.1445.

Substituting in (12) we have

$$\frac{P}{a} = 9700 \text{ pounds.}$$

from which the required area is

$$a = 0.234 \text{ square inches.}$$

Comparing this area with the area, 0.75 square inches, of the assumed member it is seen that there is a margin of safety, hence the proportions given above will be used.

GUIDE BOLTS.

The guide bolts will be considered as cantilever beams having the load uniformly distributed. The maximum bending moment then occurs at the point of support and is given by the formula

$$M = \frac{Pl}{2} \quad \dots\dots\dots\dots\dots\dots\dots \quad (15)$$

where M = maximum bending moment

P = the pressure

l = the length

Substituting the known values of P and l in (15) we have

$$M = \frac{0.75 \times 2265}{2} = 850 \text{ lb. in.}$$

Assuming a diameter of 1 1/2 inches and substituting the known values of M and d in (14) we get

$$S = 2560 \text{ lb./ sq. in.}$$

BEARING.- The bearing of the bolts must be designed for crushing and shear.

CRUSHING.- Dividing the pressure on the bolt by the projected

area we have

$$S_c = \frac{2265}{0.75 \times 1.5} = 2020 \text{ lb./ sq. in.}$$

SHEARING. - The bolts are in single shear and, hence, dividing the pressure by the area we have

$$S_s = \frac{2265}{1.767} = 1280 \text{ lb./ sq. in.}$$

These values are all within the safe limit and hence the bolts will readily carry the load coming upon them.

LOWER LINK.

For the proportions of the lower link assume those shown in Fig. 7.

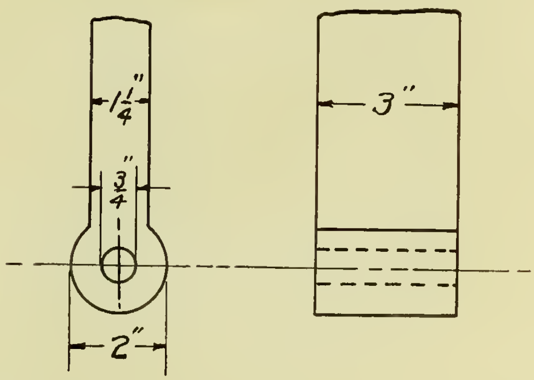


Fig. 7.

This is a short link and will therefore be designed for compression only. To determine the compressive stress in the body of the link, divide the load by the area, we have

$$S_c = \frac{72500}{3(2 - 0.75)} = 19300 \text{ lb./ sq. in.}$$

This stress appears somewhat high but is permissible.

TOGGLE LINK.

The toggle link is an eccentrically loaded member. It is in

compression, hence, a modified form of (7) can be used to calculate the stresses. The modified form of (7) is

$$S = \frac{P}{hb}\left[1 + \frac{6e}{h}\right] \quad \text{. . . . .} \quad (16)$$

where h = the depth of the link

b = the width of the link

e = eccentricity

Assuming the dimensions given in Fig. 8 below and substituting the known values of P,h,b, and e we have

$$S = 15150 \text{ lb./ sq. in.}$$

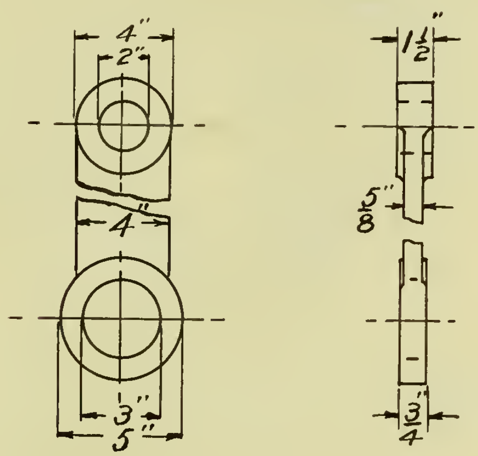


Fig. 8.

BEARINGS.- The bearings must be investigated for crushing. Therefore, dividing the total pressure by the projected area we have, for the upper end of the link

$$S_c = \frac{34700}{2 \times 1.5} = 11550 \text{ lb./ sq. in.}$$

For the lower end

$$S_c = \frac{34700}{3 \times 0.75} = 15400 \text{ lb./ sq. in.}$$

TOGGLE LEVER.

There are four forces acting on the toggle lever as shown

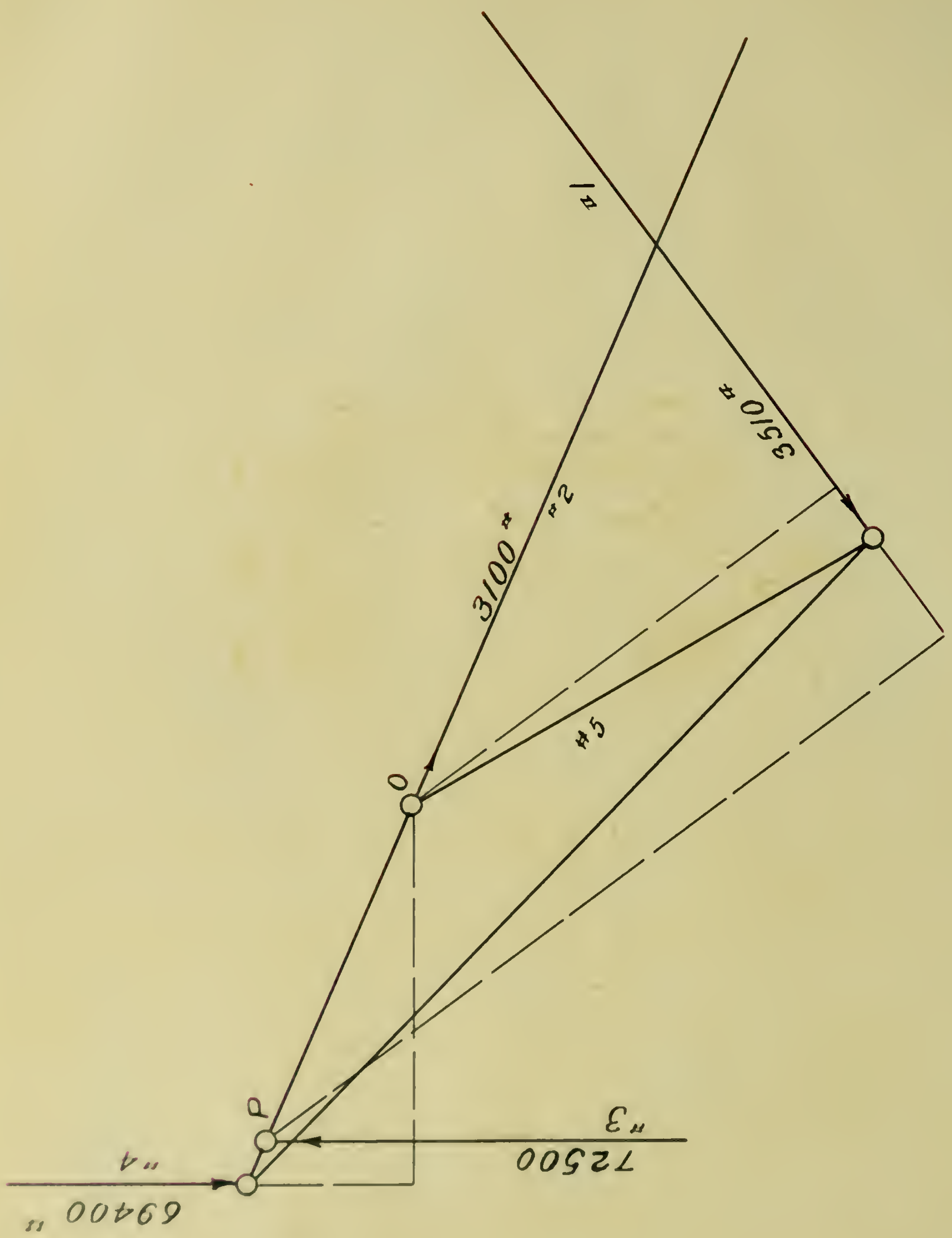


Fig. 9

in Fig. 9. In the final position of the piston it is seen that the links 2 and 3 act through the point P, hence the stress in the toggle lever may be determined by taking moments about the point P. Taking moments about the point P we have

$$M = 0.5 \times 69400 = 34700 \text{ lb. in.}$$

Assuming a section as shown in Fig. 10 and investigating the section we find that the section has the properties exhibited in Table V.

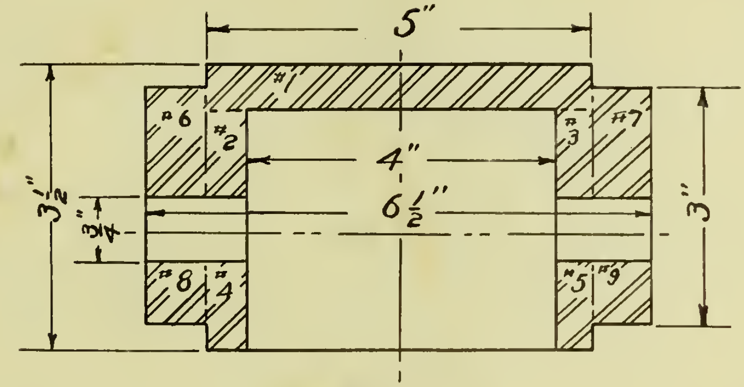


Fig. 10.

TABLE V.

No	Area	Mom. Arm.in.	Mom.	C_t	C_c	Icg	h	Ah^2	I	$\frac{I}{c}$ T_c	$\frac{I}{c}$ C_c
1	2.500	3.250	8.120	2.205	1.295	0.052	1.045	2.725	2.777	4.58	7.78
2	0.563	2.438	1.370			0.059	0.233	0.031	0.089		
3	0.563	2.438	1.370			0.059	0.233	0.031	0.089		
4	0.563	0.563	0.316			0.059	1.642	1.520	2.111		
5	0.563	0.563	0.316			0.059	1.642	1.520	2.111		
6	0.985	2.531	2.490			0.141	0.326	0.105	0.245		
7	0.985	2.531	2.490			0.141	0.326	0.105	0.245		
8	0.563	0.750	0.422			0.026	1.455	1.190	1.216		
9	0.563	0.750	0.422			0.026	1.455	1.190	1.216		
	7.848		17.316						10.099		

Substituting the proper values in (2) to (8) inclusive we have the following values. These values do not exceed the specified limit.

$$S_t = \frac{69400 \times 0.5}{4.58} = 7580 \text{ lb./ sq. in.}$$

$$S_c = \frac{69400 \times 0.5}{7.78} = 4450 \text{ lb./ sq. in.}$$

$$S_s = \frac{69400}{7.848} = 8840 \text{ lb./ sq. in.}$$

$$\text{Max. } S_t = 13410 \text{ lb./ sq. in.}$$

$$\text{Max. } S_c = 11350 \text{ lb./ sq. in.}$$

$$\text{Max. } S_s = 9640 \text{ lb./ sq. in.}$$

To find the stress at the point O, see Fig. 9 page 24, moments were taken about the point O.

$$M = 3510 \times 6.18 = 21620 \text{ lb. in.}$$

Assuming a section as shown below in Fig. 11 and investigating the section we get the values shown in Table VI.

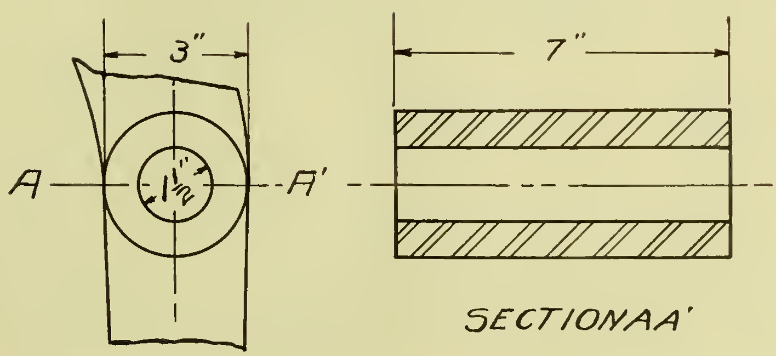


Fig. 11.

TABLE VI.

No	Area	Mom. Arm. in.	Mom.	C_t	C_c	Icg	h	Ah^2	I	$\frac{I}{c}$ T_c	$\frac{I}{c}$ C_c
1	5.25	0.375	1.97	1.5	1.5	0.23	1.125	6.65	6.88	9.16	9.16
2	5.25	2.625	13.80			0.23	1.125	6.65	6.88		
	10.50		15.77						13.76		

Substituting the values given in Table VI in (14) we have

$$S_c = S_t = \frac{21620}{9.16} = 2360 \text{ lb./ sq. in.}$$

$$S_s = \frac{21620}{10.5} = 2250 \text{ lb./ sq. in.}$$

These values are low but a wide section must be used on account of clearances.

The toggle lever arms will be considered as cantilever beams uniformly loaded. Hence substituting the known values of P and l in (15) we have for the maximum bending moment at the point of support

$$M = \frac{34700 \times 0.75}{2} = 13000 \text{ lb. in.}$$

Substituting the known values of M, and d in (14) we have

$$S = \frac{32 \times 13000}{27} = 4920 \text{ lb./ sq. in.}$$

which is a safe value to use.

TOGGLE BOLT.

The toggle bolt will be considered as a beam fixed at both ends and carrying two concentrated loads as shown in Fig. 12.

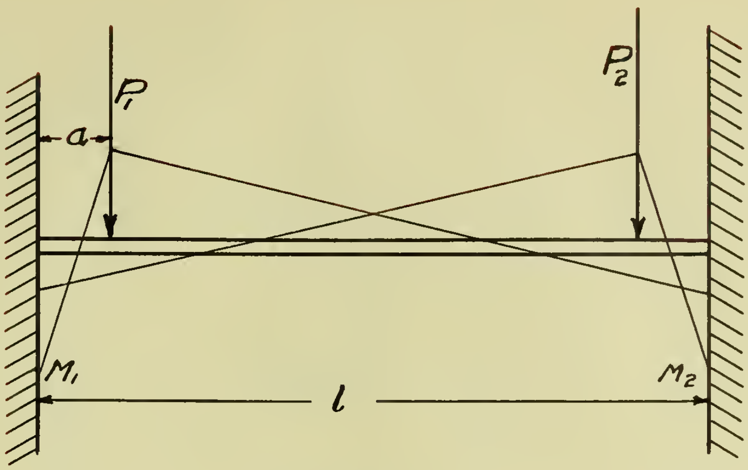


Fig. 12.

The formulae for a beam fixed at both ends and carrying one concentrated load are

$$M_1 = -Pa\left[1 - \frac{2a}{l} + \frac{a^2}{l^2}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (17)$$

$$M_2 = -\frac{Pa^2}{l}\left[1 - \frac{a}{l}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (18)$$

$$M_a = +P\frac{a^2}{l}\left[2 - \frac{4a}{l} + \frac{2a^2}{l^2}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \quad (19)$$

where M_1 = the bending moment at the left support.

M_2 = the bending moment at the right support.

M_a = the bending moment under the load.

P = the load.

l = the distance between supports.

a = the distance of the load from the left support.

Substituting the known values of P, a, and l in (17), (18), and (19) we have

$$M_1 = -15150 \text{ lb. in.}$$

$$M_2 = - 2490 \text{ lb. in.}$$

$$M_a = + 4460 \text{ lb. in.}$$

These values give the bending moments due to one load but since the beam carries two symmetrical loads it is evident that, due to P_2, there will be a bending moment M_2' equal to M_1 at the right support another bending moment M_1', equal to M_2, at the left support. The maximum bending moment will evidently occur at one of the supports and is equal to the algebraic sum of the two moments at the support, or, since the loads are symmetrical

$$M = M_1 + M_2 = - 15150 - 2490$$

$$= - 17640 \text{ lb. in.}$$

Assuming a diameter of 2 inches and substituting the known values of M and d in (14) we have

$$S = \frac{32 \times 17640}{8} = 22500 \text{ lb./ sq. in.}$$

BEARING.- The bolt must be investigated for crushing and shear, hence, dividing the pressure by the area we have for the stresses

$$S_c = \frac{34700}{1.5 \times 2} = 11560 \text{ lb./ sq. in.}$$

$$S_s = \frac{34700}{3.14} = 11000 \text{ lb./ sq. in.}$$

These values are safe for a toggle bolt, the relative motion between the surfaces in contact is slight and the speed is slow.

TOGGLE BOLT SUPPORT.

This is the top part of the frame and must be designed to take up the reaction of the plunger. It is eccentrically loaded, hence, assuming a section as shown in Fig. 13 and investigating the section, we find that the section has the properties tabulated

in Table VII.

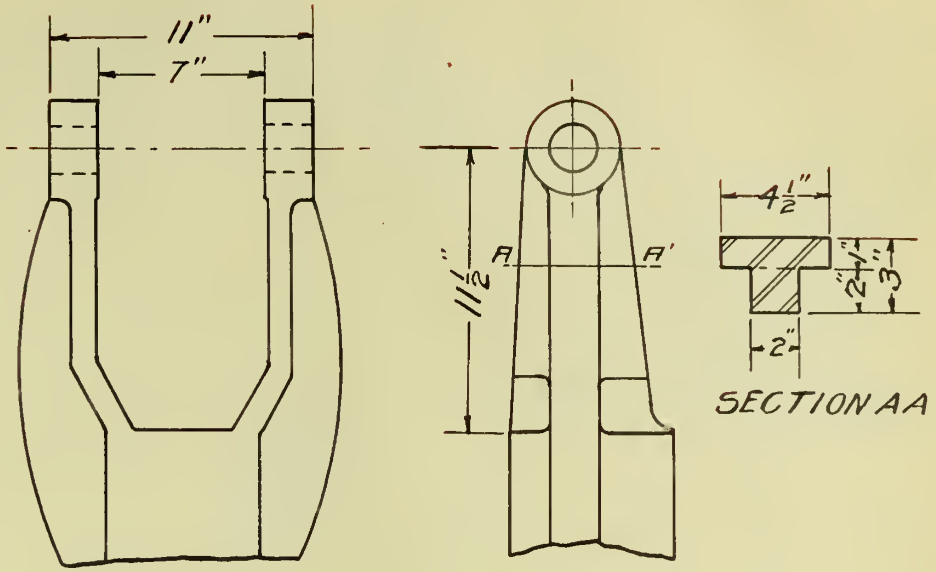


Fig. 13.

TABLE VII.

No	Area	Mom. Arm. in.	Mom.	C_t	C_c	Icg	h	Ah^2	I	$\frac{I}{c}$ T_c	$\frac{I}{c}$ C_c
1	4.5	0.5	2.25	1.205	1.795	0.375	0.705	2.24	2.615	5.37	3.60
2	4.0	2.0	8.00			1.330	0.795	2.51	3.840		
	8.5		10.25						6.455		

Substituting the values of Table VII in (2) to (8) inclusive we have

$$P = 34700 \text{ lb.} \qquad e = 1.955 \text{ in.}$$

$$S_t = \frac{34700 \times 1.955}{5.37} = 12650 \text{ lb./ sq. in.}$$

$$S_c = \frac{34700 \times 1.955}{3.6} = 18800 \text{ lb./ sq. in.}$$

$$S_s = \frac{P}{a} = \frac{34700}{8.5} = 4080 \text{ lb./ sq. in.}$$

$$\text{Max. } S_t = 15500 \text{ lb./ sq. in.}$$

$$\text{Max. } S_c = 19650 \text{ lb./ sq. in.}$$

These values are within the specified limits.

The parts of the riveter for which no calculations have been shown have been designed from judgement; for instance the cylinder ports were designed so they would cast well. The valve was designed for a good wearing surface, and the lever used for operating the valve was proportioned so it would look well when compared with the control lever.

BILL OF MATERIAL.

One of the features of modern manufacturing is the standardization of parts so that repairs may be sent out from the factory without delay. In order to do this successfully it is necessary to have a system of symbols, or numbers, so that parts of a machine may be easily distinguished. In fact, where a good system is in use the actual name of the part is seldom referred to but the piece is known by its symbol.

In this thesis the following system is used. Each piece is given a symbol and a number. The symbol refers to the material that the part is made of and the number distinguishes it from other parts of the same material. Thus A means that the part is made from steel casting, B from cast iron, etc. A2 is then a particular piece made from cast steel, B2 is some other part made from cast iron, etc.

The Bill of Material is divided into two parts; Manufactured Parts and Standard Parts. The manufactured parts are those which are actually made in the factory, or require finishing before being assembled, while the standard parts are such as bolts, nuts, set screws, or any part that can be purchased ready for assembling.

In the following "Bill of Material" are listed all parts, whether manufactured or standard, that are used in the building of this pneumatic riveter.

BILL OF MATERIAL.

Manufactured Parts.

No.	Name of Parts.	Material	Pat. No.	No. Req'd.
A1	Frame	Steel Casting.	A1	1
A2	Plunger	"	A2	1
A3	Toggle Link	"	A3	2
A4	Connecting Rod	"	A4	1
A5	Lower Toggle	"	A5	1
A6	Connecting Rod Clevice	"	A6	1
A7	Toggle Lever	"	A7	1
B1	Cylinder	Cast Iron	B1	1
B2	Piston	"	B2	1
B3	Cylinder Back Head	"	B3	1
B4	Cylinder Front Head	"	B4	1
B5	Valve Bonnet	"	B5	1
B6	Valve	"	B6	1
B7	Spacing Collar	"	B7	1
C1	Toggle Bolt	Steel.		1
C2	Toggle Lever Washer	"		2
C3	Valve Lever	"		1
C4	Guide Link	"		2
C5	Adjusting Screw	"		1
C6	Guide Bolt	"		2
C7	Clevice Pin	"		1
C8	Piston Pin	"		1
C9	Packing Ring	"		2
C10	Toggle Pin	"		1

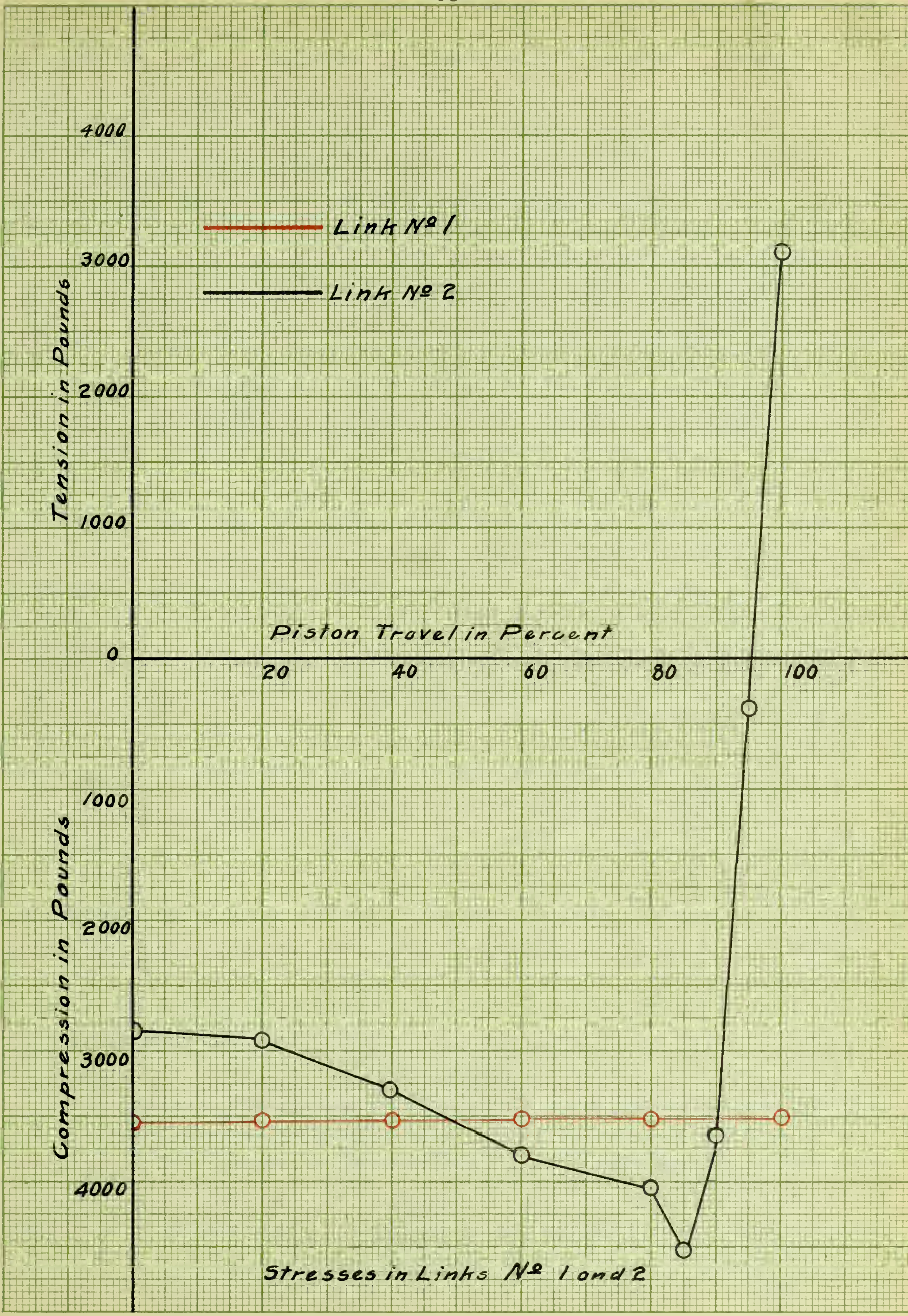
BILL OF MATERIAL.

Manufactured Parts.

No.	Name of Parts.	Material	Pat. No.	No. Req'd.
C11	Guide Washer	Steel		4
C12	Valve Stem Washer	"		2
C13	Frame Bolts	"		6
C14	Plunger Pin	"		1
C15	Valve Stem	"		1
C16	Control Lever	"		1
D1	Link Block	Bronze.		1
D2	Valve Gland	"		2
E1	Non-Metalic Packing			2
E2	Leather Packing			1

Standard Parts.

Name of Parts.	No. Req'd.
1/2" x 1" U.S.S. Set Screw	2
1/2" x 1 1/4 U.S.S. Set Screw	1
3/8" U.S.S. Nut	4
1/2" U.S.S. Nut	8
5/8" U.S.S. Nut	10
3/4" U.S.S. Nut	6
1" U.S.S. Nut	7
1 1/2" U.S.S. Nut	1
5/16" Taper Pin	1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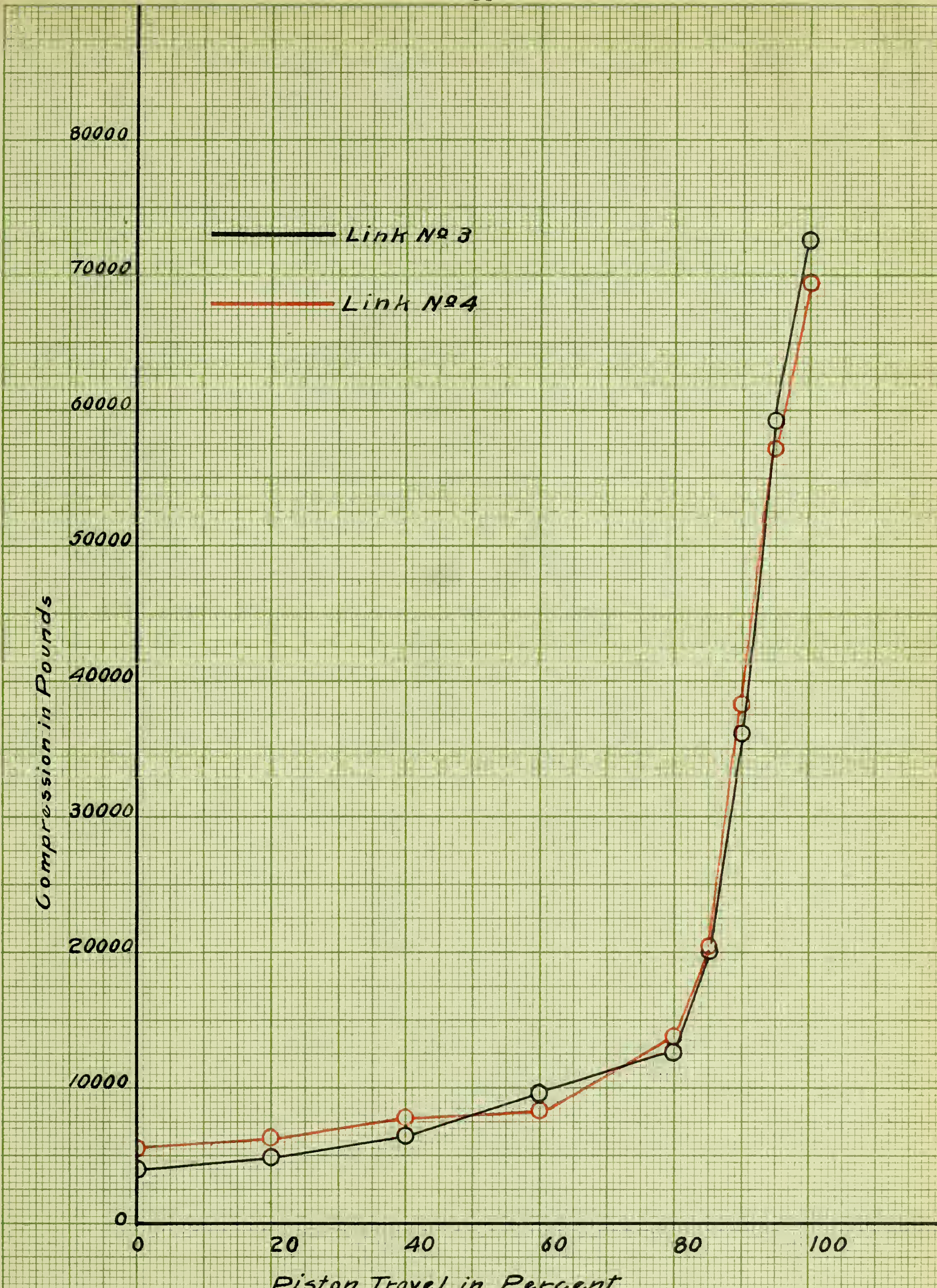
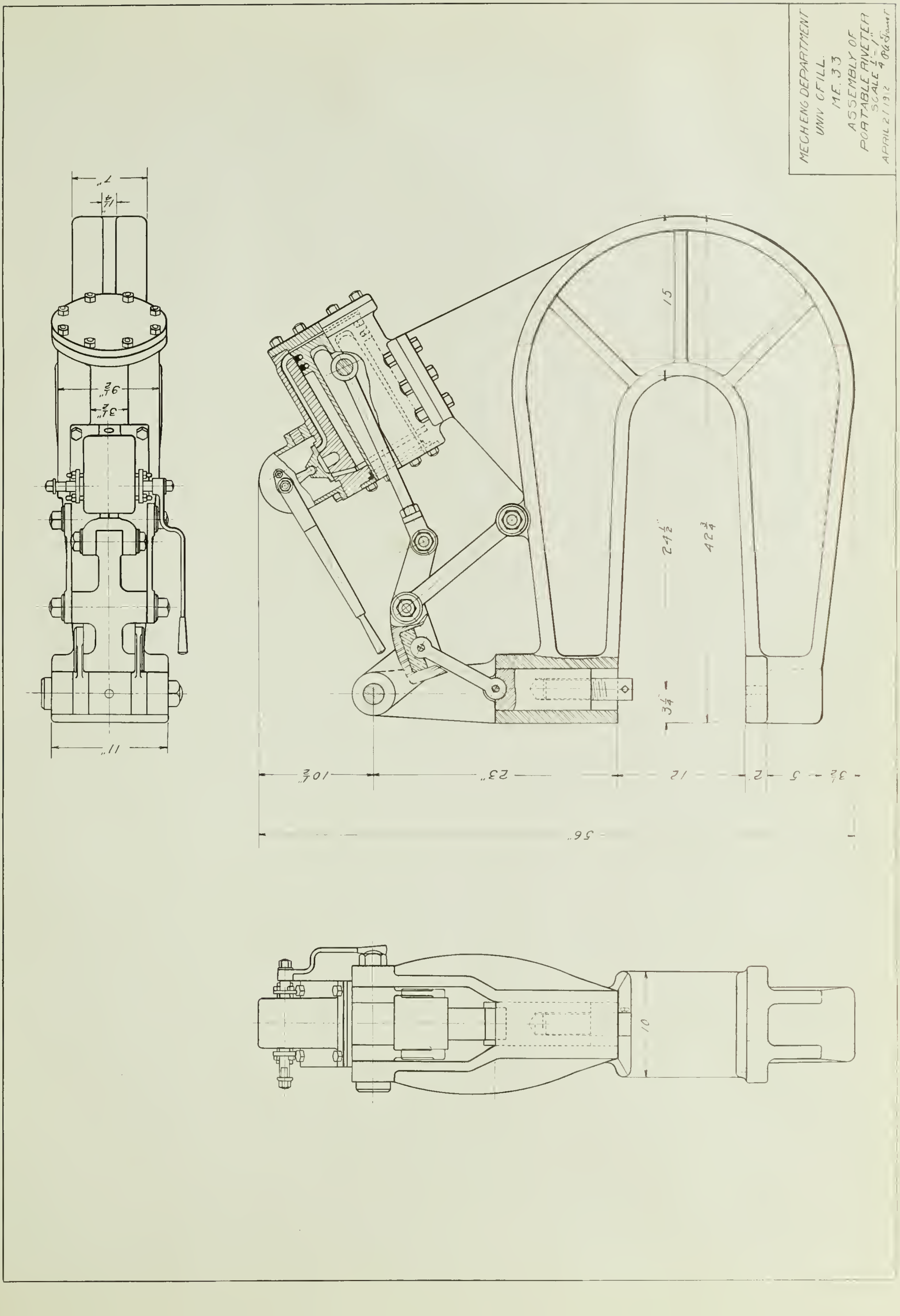
Link No 2
Tension in Pounds
4000
3000
2000
1000
0
Piston Travel in Percent
20
40
60
80
100
1000
2000
3000
4000
Compression in Pounds
Stresses in Links No 1 and 2
NO 346
EUGENE DIETZGEN



Stresses in Links Nº 3 and 4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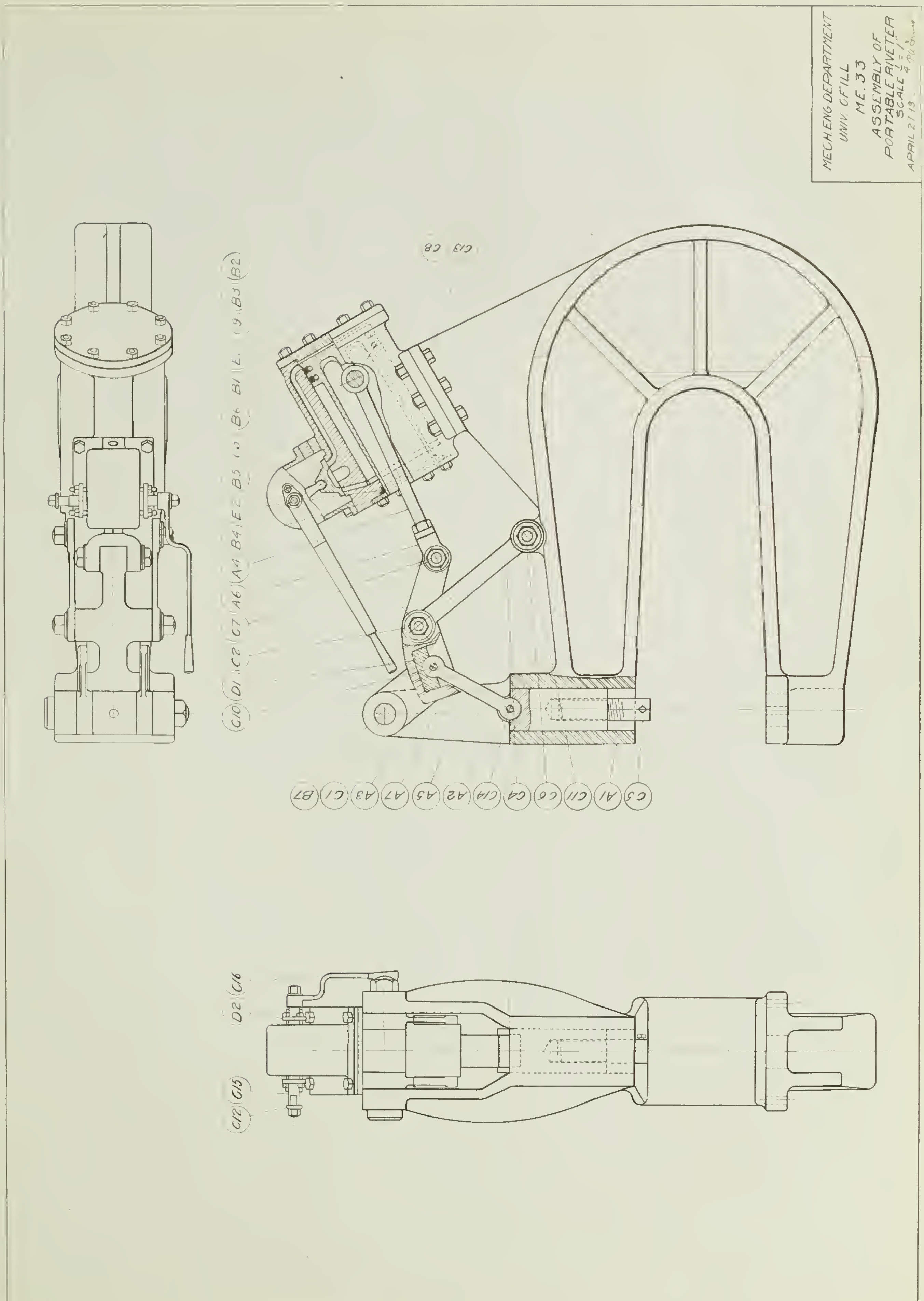
UNIV OF ILL.
M.E. 33
ASSEMBLY OF
PORTABLE RIVETER
SCALE 1/4" = 1"
APRIL 21 1912
7"
1/4"
9 1/2"
3 1/2"
11"
15
24 1/2"
42 3/4
3 1/4
10 1/2"
23"
12
2
5
3 1/2
56"
10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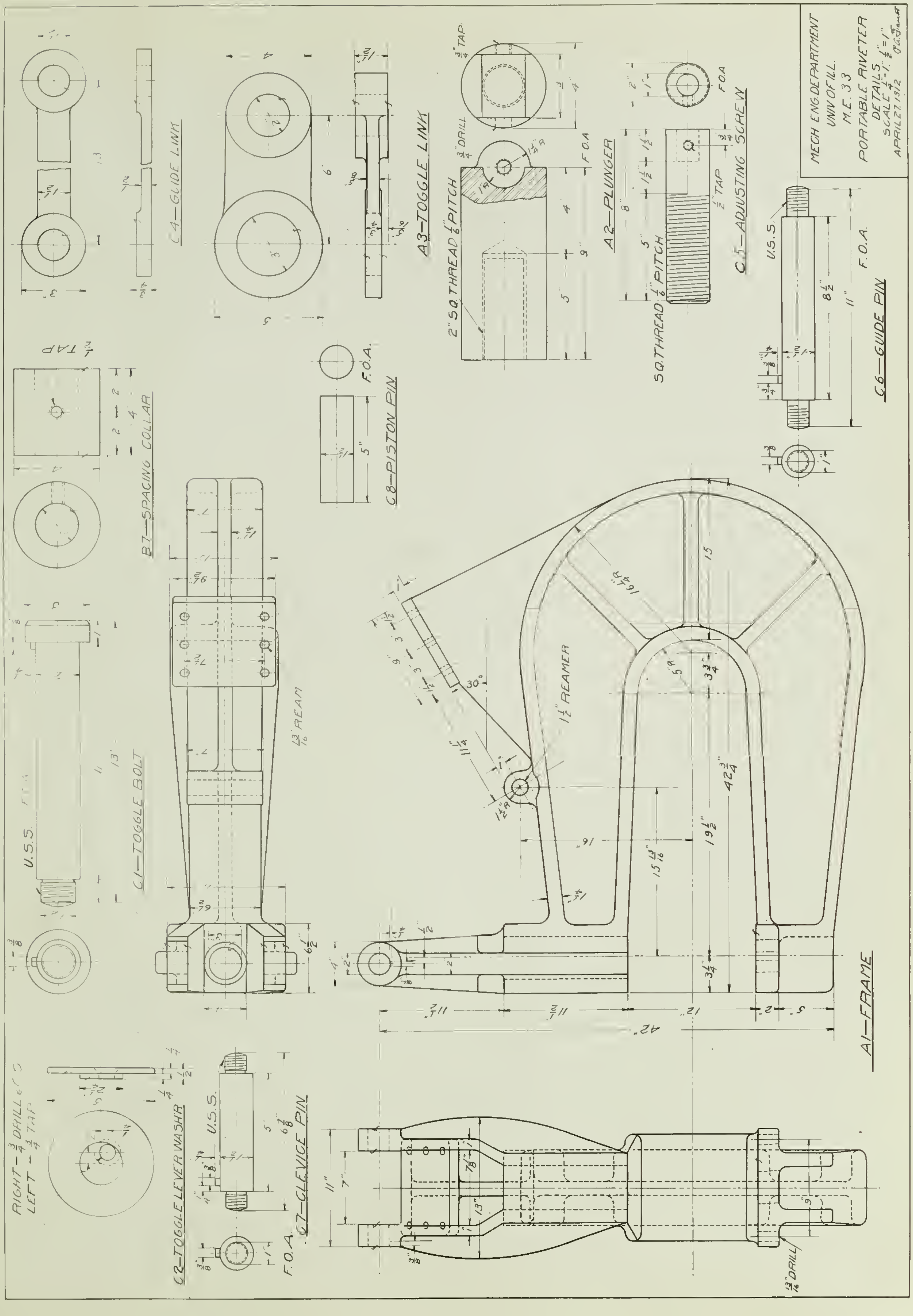
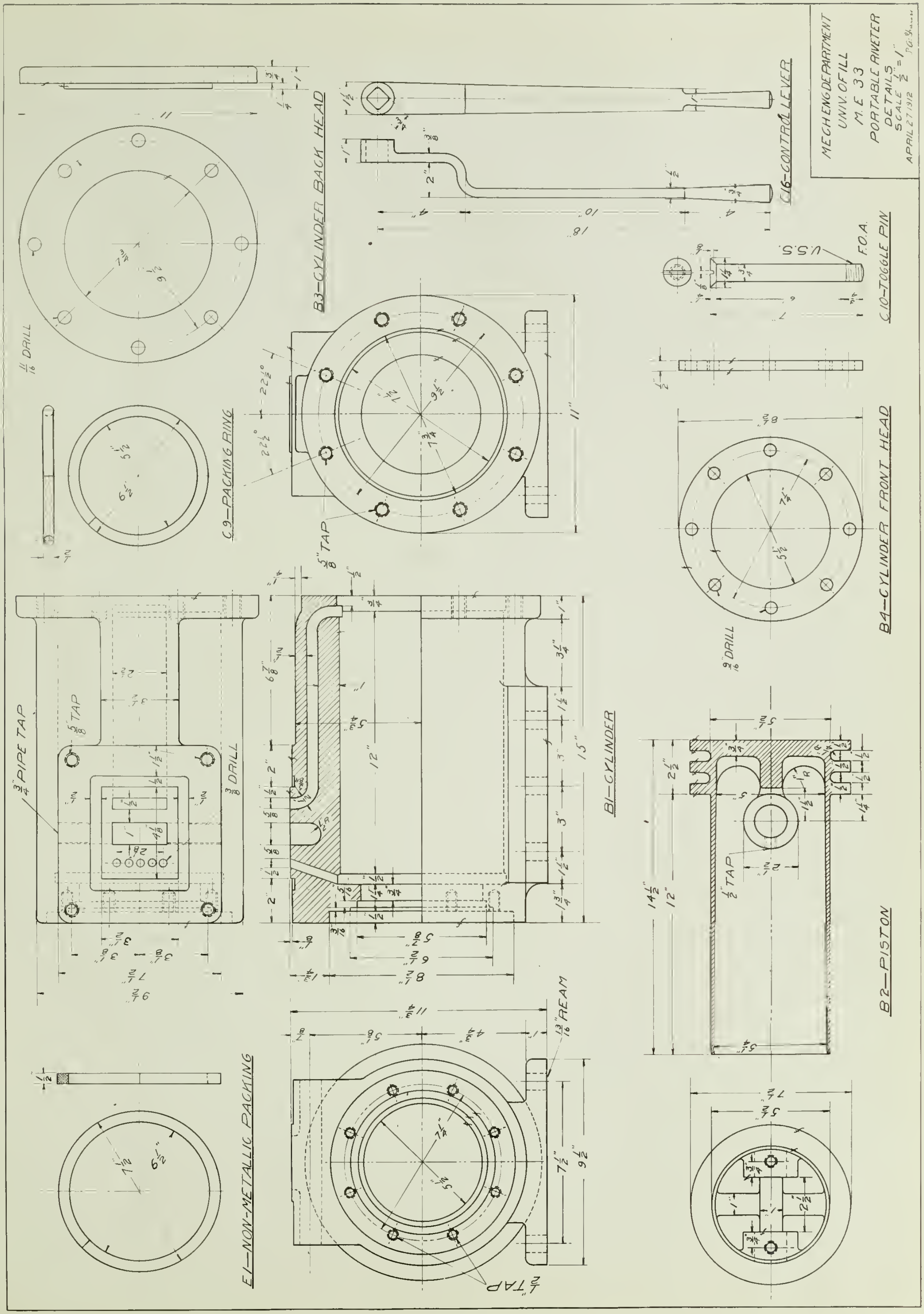
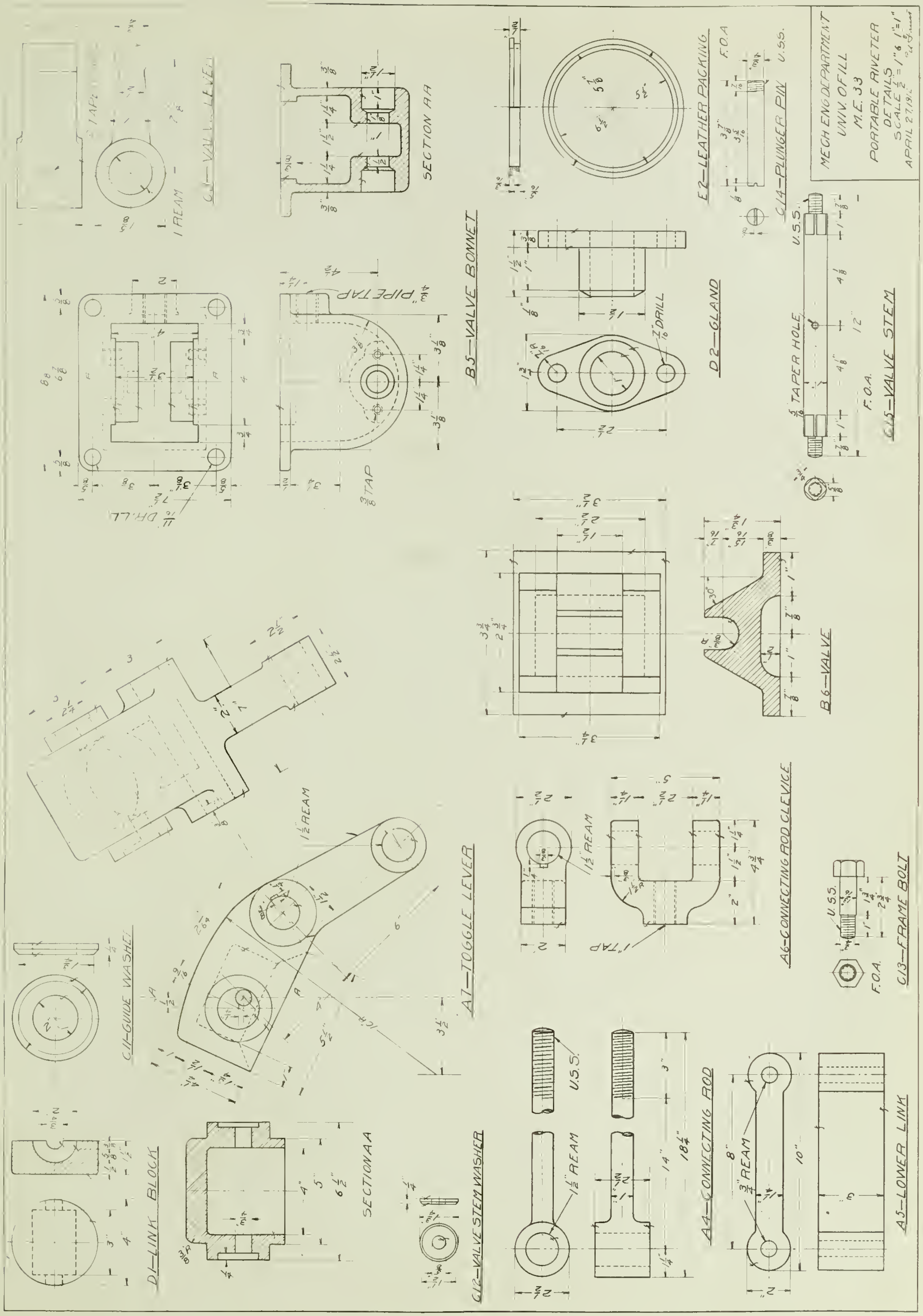
UNIV. OF ILL.
M.E. 33
ASSEMBLY OF
PORTABLE RIVETER
SCALE 1/4"=1"
APRIL 2/19.
C10 D1 C2 C7 A6 A4 B4 E2 B5 C3 B6 B1 E1 C9 B3 B2
C13 C8
C5 A1 C11 C6 C4 C14 A2 A5 A7 A3 C1 B7
D2 C16
C12 C15


MECH. ENG. DEPARTMENT
UNIV. OF ILL.
M.E. 33
PORTABLE RIVETER
DETAILS
APRIL 27, 1912
A1—FRAME
A2—PLUNGER
A3—TOGGLE LINK
B7—SPACING COLLAR
C1—TOGGLE BOLT
C2—TOGGLE LEVER WASH'R
C4—GUIDE LINK
C5—ADJUSTING SCREW
C6—GUIDE PIN
C7—CLEVICE PIN
C8—PISTON PIN
2" SQ. THREAD 1/6" PITCH
SQ. THREAD 1/6" PITCH
U.S.S.
F.O.A.
1 1/2" REAMER
13/16" REAM
13/16" DRILL
3/4" DRILL
3/4" TAP
1/2" TAP
RIGHT–3/4 DRILL
LEFT–3/4 TAP


MECH ENG DEPARTMENT
UNIV. OF ILL
M.E 33
PORTABLE RIVETER
DETAILS
SCALE 1/2" = 1"
APRIL 27 1912
B3—CYLINDER BACK HEAD
C16—CONTROL LEVER
C10—TOGGLE PIN
U.S.S.
F.O.A.
11/16 DRILL
C9—PACKING RING
5/8" TAP
B4—CYLINDER FRONT HEAD
9/16 DRILL
3/4" PIPE TAP
5/8" TAP
3/8 DRILL
B1—CYLINDER
1/2" TAP
B2—PISTON
13/16" REAM
E1—NON-METALLIC PACKING
1/4" TAP


MECH ENG DEPARTMENT
UNIV. OF ILL
M.E. 33
PORTABLE RIVETER
DETAILS
D1—LINK BLOCK
SECTION AA
C11—GUIDE WASHER
C1—VALVE LEVER
C12—VALVE STEM WASHER
A7—TOGGLE LEVER
B5—VALVE BONNET
SECTION AA
A4—CONNECTING ROD
A6—CONNECTING ROD CLEVICE
B6—VALVE
D2—GLAND
E2—LEATHER PACKING
C14—PLUNGER PIN
C15—VALVE STEM
C13—FRAME BOLT
A5—LOWER LINK